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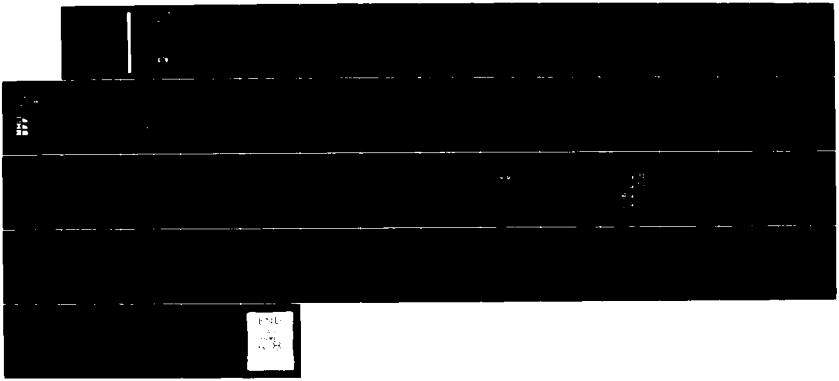


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Characterization of lubricants in terms of transition diagram data

Final Report

by

J.W.M. Mens and A.W.J.de Gee

March 1988

United States Army

EUROPEAN RESEARCH OFFICE OF THE U.S. ARMY

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TNO Metals Research Institute, Apeldoorn, The Netherlands

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Characterization of lubricants, meant for application in counterformal contact situations (concentrated sliding contacts), is usually performed with standard tests, e.g. of the four-ball type. Results, thus produced, are of limited value, because of doubts, regarding correlation with practical performance data. In this respect the IRG (International Research Group) Transition Diagram method, developed by the co-operative efforts of eight different laboratories, probably stands a better chance of success. As an example, five high-performance lubricants i.e. three mineral oils, i.e. Amoco 300, Citgo C-5 and

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19. REO 203 and two synthetic oils, i.e. Mobil Jet II and Royco 555, were characterized with this method. Thereby the following quality criteria were applied:

- load carrying capacity of the (partial) EHD film at sliding speeds v of 0.4 m/s (I-II transition) and 4 m/s (I-III transition).
- load carrying capacity of the boundary lubricant film at v = 0.4 m/s (II-III transition).
- friction-time and wear-time curves, when operating under conditions of boundary lubrication (lubrication regime II).

Results, obtained with hardened (H = 8 GPa) ball bearing steel specimens at 60°C oil bath temperature, show that quality ranking depends on the test criterion, particularly as far as the difference between mineral and synthetic lubricants is concerned. The same applies to correlation with standard tests, indicating the arbitrary character of the latter. At v = 0.4 m/s the synthetic oils perform (much) better than the mineral oils (notwithstanding their much lower viscosity), particularly with regard to the load carrying capacity of the boundary lubricant film. This is due to a significant reduction in friction and, thus, in heat production in the contact zone. At v = 4 m/s excellent correlations with four-ball test results are obtained. Results of tests with as-machined surfaces correlate with four-ball 2.5 s seizure load results; results of tests with run-in surfaces correlate with four-ball Load Wear Index and Weld Point Load.

Additional test series were performed with softer (H = 3 GPa) specimens and at higher (120°C and 180°C) oilbath temperature.

Using relatively soft specimens leads to severe plastic deformation, prior to transition. Invariably, I-III transitions occur, i.e. at v = 4 m/s as well as at v = 0.4 m/s. At these speeds different rankings are obtained, which also differ from any of the rankings produced before (i.e. at H = 8 GPa). Under conditions of severe plastic deformation Royco oil is found to behave exceptionally well, also at v = 4 m/s!

With Mobil (tested at T = 60, 120 and 180°C) and Amoco (tested at T = 60 and 120°C) the load carrying capacity F_c generally decreases with increasing oil bath temperature T. On the contrary the F_c-values of surfaces, lubricated with Royco (tested at 60, 120 and 160°C) remain roughly the same (at H = 8 GPa) or increase appreciably with increasing T (at H = 3 GPa).

CHARACTERIZATION OF LUBRICANTS IN TERMS OF TRANSITION DIAGRAM DATA

Abstract

Characterization of lubricants, meant for application in counterformal contact situations (concentrated sliding contacts), is usually performed with standard tests, e.g. of the four-ball type. Results, thus produced, are of limited value, because of doubts, regarding correlation with practical performance data. In this respect the IRG Transition Diagram method, developed by the co-operative efforts of eight different laboratories, probably stands a better chance of success. As an example, five high-performance lubricants i.e. three mineral oils, i.e. Amoco 300, Citgo C-5 and REO 203 and two synthetic oils, i.e. Mobil Jet II and Royco 555, were characterized with this method. Thereby the following quality criteria were applied:

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List of keywords
Counterformal contacts, lubricant characterization, IRG Transition Diagram, Hardness effects, ASTM Standard tests.

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Lubricant : Mobil (nr. 5)
v : 0.4 m/s
F : 500, 600 and 650 N
H : 3 GPa

Fig.14 Transition forces F_c at $T = 60^{\circ}C$ and $H = 3$ GPa.

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T = 60, 120 and 180°C
V = 0.4 m/s
F = 700 N

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Lubricants: Mobil (M) and Royco (R).
T = 60, 120 and 180°C
V = 0.4 m/s
F = 700 N

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Lubricants : Mobil (M) and Royco (R).
Test conditions: see caption Fig. 18.

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<u>Table 14</u> Influence of oilbath temperature T on the F_c-values of Amoco.
$H = 8$ GPa.

List of Symbols

d	:	sliding distance	(km)
d^0	:	running-in distance	(m)
D_E	:	diameter of (nearly) circular contact surface, formed by elastic deformation	(um)
$D_W^{\,0}$	:	diameter of wearscar, formed during running-in	(um)
f	:	coefficient of friction	(-)
F	:	normal force	(N)
F_c	:	transition force	(N)
F_F	:	friction force	(N)
k	:	specific wearrate	(10^{-6} mm^3/Nm)
$\bar{p}_c$	:	average transition pressure	(GPa)
t	:	time	(s)
T	:	oilbath temperature	(^{0}C)
T_c	:	critical temperature I-II transition	(^{0}C)
v	:	*sliding speed*	(*m/s*)
ΔV^0	:	volume wear due to running-in	(10^{-3} mm^3)
ΔV_{15}	:	volume wear after 15 km sliding in regime II	(10^{-2} mm^3)
γ	:	correlation coefficient for linear regression $0.90 < \gamma < 1.00$: positive correlations $-1.00 < \gamma < -0.90$: negative correlations.	(-)
υ	:	kinematic viscosity	(mm^2/s)

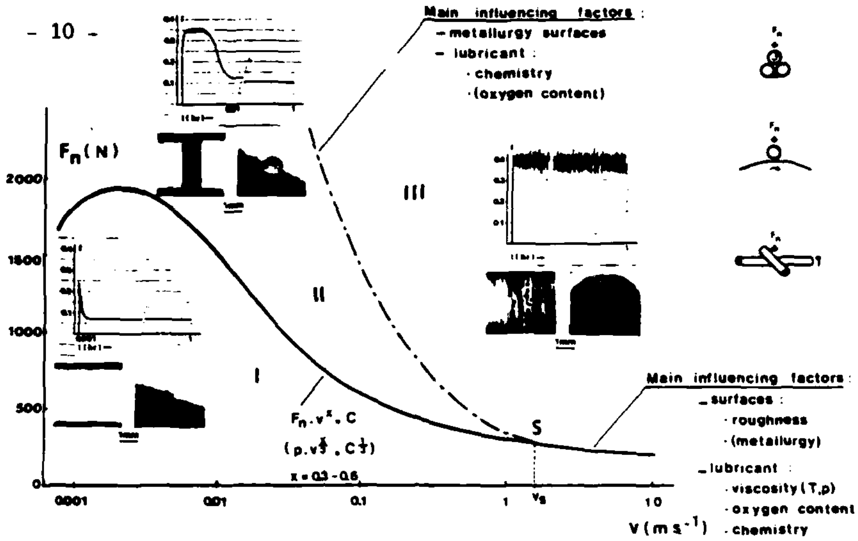


Fig. 1 Cross section at constant oil bath temperature through an IRG transition diagram for fully oil-submerged sliding concentrated contacts (schematic presentation; see Fig. 1A for magnifications of inserts).

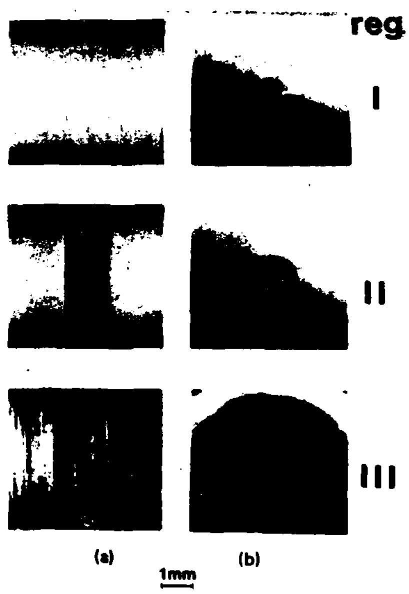


Fig. 1A A: Cylinder surfaces
B: Ball surfaces.
The black shadows, visible on the ball surfaces, are caused by the photographic illumination technique.

1 OBJECTIVE

The objective of the work was to give a comprehensive functional characterization of five lubricants, provided by the US Army Fuel and Lubricants Research Laboratory.

2 EXPERIMENTAL TECHNIQUES

2.1 The IRG Transition Diagram Technique

For the greater part the tests were based on the transition diagram technique, developed in co-operative research by the International Research Group on Wear of Engineering Materials IRG-OECD. In this method, which has been described comprehensively in [1], the condition of lubrication (i.e. partial elastohydrodynamic, boundary or 'unlubricated') of fully oil-submerged initial point or line contacts is described as a function of normal force on the contact F, relative sliding speed v and oil bath temperature T. A cross section at constant T of such a transition diagram is shown in Fig. 1. In the diagram the lubrication regimes I (partial elastohydrodynamic), II (boundary lubrication) and III (virtually unlubricated) are separated by 'transition curves', the location of which depend on a number of parameters, as indicated in Fig. 1 and discussed in detail in ref. 1.

In the present case the transition diagram tests were performed with a TNO tribometer in which a stationary ball with Ø = 10 mm is pressed against the curved surface of rotating ring with Ø = 73 mm. A schematic presentation of the essential parts of the equipment is shown in Fig. 2. Tests were performed at sliding speeds v = 0.4 and 4 m/s. Generally, testing at v = 0.4 m/s at increasing F-values leads to I-II and II-III transitions, i.e. the tests are performed at $v < v_s$ (c.f. Fig. 1). At v = 4 m/s only I-III transitions occur, i.e. $v > v_s$.

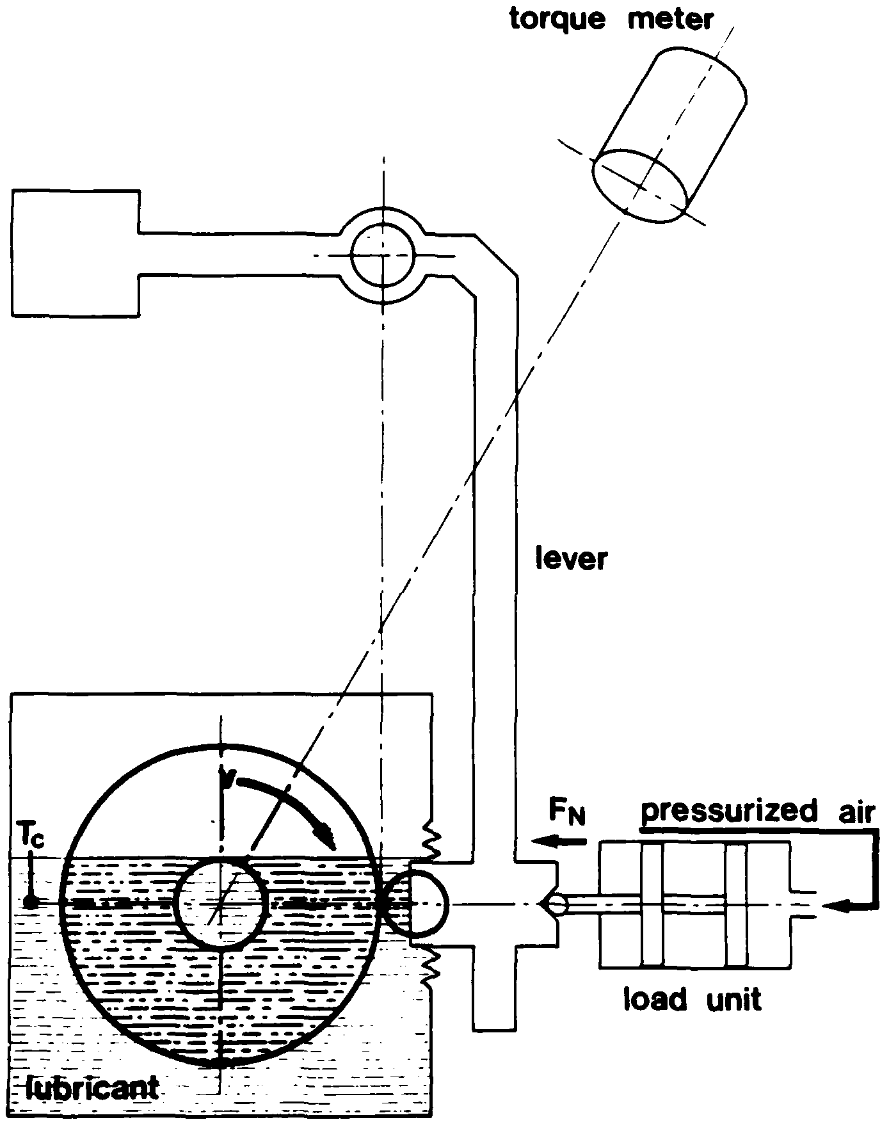


Fig. 2 Essential parts of TNO tribometer.

In testing virginal or run-in surfaces (see below), separate two min. tests were performed at increasing F values, until a transition from one lubrication regime to another was found (transition force F_c, determined with an accuracy of $\pm$ 25 N). For the I-II transitions, occurring at $v = 0.4$ m/s, the main transition indicator was the coefficient of friction f (c.f. Fig. 1; f-t diagram inserts); for the II-III transition at $v = 0.4$ m/s and the I-III transition at $v = 4$ m/s it was the wearrate (when operating in regime III, about 25% of the stationary (steel) ball wears away during a two min. test!).

Tests were performed with virginal and with run-in surfaces. In the latter case running-in procedure B, described in [2] was applied. To that purpose the surfaces were run together under a normal force $F^o = 150$ N for 1000 s (53 revolutions), at a running-in speed $v^o = 0.1$ m/s. The running-in distance d^o thus amounted to 100 m. Under such conditions the system operated in lubrication regime I (partial elastohydrodynamic) and a small amount of mild 'running-in wear' occurred.

2.2 Practical significance of transition force determinations

The results of transition force determinations relate to cams and tappets and gears. In testing virginal (not run-in) surfaces, the test results relate to practical situations, in which virginal parts of the contacting bodies (e.g. gearteeth) suddenly come into contact, for instance as a result of shaft deflections, due to shock loading. If testing run-in surfaces, the test results relate to components, running under stationary contact conditions. Under conditions of severe plastic deformation ($H = 3$ GPa) the results relate primarily to metal forming operations.

2.3 Long-term tests under conditions of (initial) boundary lubrication

In addition to the above F_c-determinations, long-term (10 h) tests under conditions of initial boundary lubrication at $v = 0.4$ m/s were performed (i.e. regime II in Fig. 1). As has been pointed out previously [3], in comparing different lubricants with respect to their wear mitigating effect when operating under conditions of boundary lubrication, one has to make sure that true boundary lubrication conditions prevail in all cases, i.e. determination of I-II and II-III transitions for all lubricants must precede long-term testing.

2.4 ASTM Standard tests

For reasons of comparison, the results of Timken and Four-Ball tests were made available by the US Army Fuels and Lubricants Research Laboratory. In addition Four-Ball 2.5 Seizure Loads $F_{c2.5}$ and Weld Point Loads (WPL) were determined at the Laboratory for Machine components and Tribology of Delft University of Technology.

3 MATERIALS AND LUBRICANTS

All tests were performed with balls and rings, made from ball bearing steel AISI 52100. Balls and rings were commercially available components of rolling element bearings with roughness values R_a of 0.01 um and 0.12 um c.l.a., respectively.(Note that in application-directed testing the contacting bodies should be made from the materials (to be) applied in practice, in which case a crossed cylinder geometry is usually to be preferred.

Part of the tests were performed with specimens in the as-delivered condition, i.e. with a Vickers hardness of 8 GPa. In addition, tests with heat-treated specimens, with a Vickers hardness of 3 GPa, were performed.

Three mineral and two fully synthetic lubricants were included in the test. These are listed in Table 1. It can be seen that they represented two groups of (kinematic) viscosity, i.e. ν (60°C) = 44-66 mm²/s (mineral lubricants) and ν (60°C) = 14-15 mm²/s (synthetic lubricants).

4 SURVEY OF TEST PROGRAMS

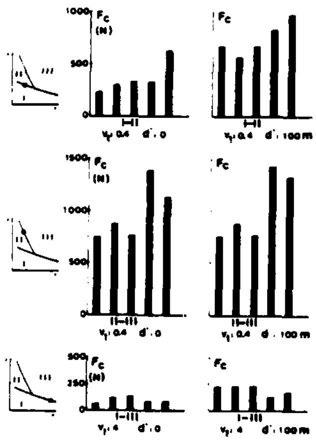
Tests were performed at three values of oilbath temperature T, i.e. 60, 120 and 180°C and two values of steel hardness, i.e. H = 8 GPa and H = 3 GPa. Table 2 gives a survey of the test programs. It can be seen that from the 30 possible combinations of lubricant, oilbath temperature and steel hardness only 17 combinations were characterized.

5 MUTUAL COMPARISON OF THE FIVE LUBRICANTS AT T = 60°C AND T = 8 GPa (TEST SERIES 1-5)

5.1 Wear during running-in

In Table 3, column 2, the results of wearscar measurements, performed after termination of the running-in periods, but before the actual transition load determinations, are given. It can be seen that wearscars with diameters, ranging from 650 um to 775 um develop. These correspond to amounts of volume wear ΔV^0, ranging from 1.6 to 3.1 . 10^{-3} mm^3. When a 'regime I force' is applied on a point contact, a brief high friction period ($f_{max} \approx 0.2$) occurs (see left insert in Fig. 1), lasting some 5-10 revolutions. During this 'high-friction' period, the specific wearrate k amounts to 1-10 . 10^{-6} mm^3/Nm [4]. At attaining the low-friction ($f < 0.1$) level, characteristic for lubrication regime I, k assumes a value of the order of 0.01 - 0.1 . 10^{-6} mm^3/Nm and - although it may take a considerable period of time (i.e. a considerable sliding distance) for wear to become entirely nil - it gradually reduces to zero at prolonged running in regime I (establishment of full elastohydrodynamic lubrication). The present ΔV values - measured after 53 revolutions - correspond to average k values, ranging from 0.11 to 0.21 . 10^{-6} mm^3/Nm, which agrees well with the characteristic ranges of k-values, given above.

The differences in ΔV values (or in k values) are characteristic for the differences in wear mitigating action of the different lubricants, when operating under conditions of partial elastohydrodynamic lubrication.



<u>Fig. 3</u> Transition forces F_c, at T = 60°C and H = 8 GPa.

It can be concluded that - at least in this respect - the group of synthetic oils (oils 4 and 5) does not distinguishes itself systematically from the group of mineral oils (oils 1-3).

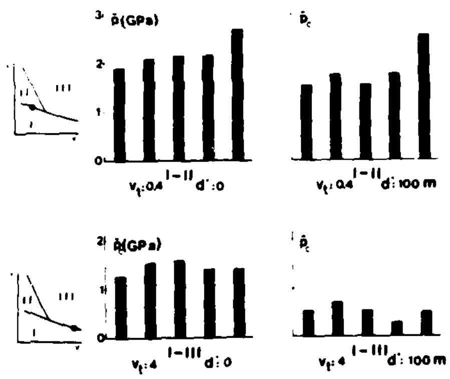
Clearly, when performing tests with run-in surfaces, virginal parts of the surfaces should not come into contact upon application of the test loads, applied at speeds v of 0.4 and 4 m/s . This means that - in all cases - the diameter D_E of the (nearly) circular contact surface, which would form on virginal (as machined) test surfaces by elastic deformation under the applied test load, should be smaller than or equal to the diameter D_w^0 of the scar, formed as a result of running-in wear. To verify this, the D_E values, calculated by application of Hertz' equations on the basis of transition force F_c values, measured upon performing tests with run-in surfaces (Table 4), are given in columns 4, 5 and 6 of Table 3. It can be seen that in all cases the above requirement is met (i.e. $D_E \leq D_w^0$).

5.2 Transition forces and transition pressures

Transition force F_c values are shown in Table 4 and in Fig. 3 for virginal as well as for run-in surfaces. These F_c values will be discussed in regard to 1) the effect of running-in and 2) the effect of lubricant composition.

5.2.1. Effect of running-in

Fig. 3 shows that running-in has a well-marked beneficial influence on the transition forces for the I-II transition (Fig. 3; top diagrams) and the II-III transition (Fig. 3; bottom diagrams), i.e. on the load carrying capacity of the partial elastohydrodynamic lubricant film. As far as the quality ranking of the lubricants is concerned, the effect of running-in is particularly pronounced in the case of Amoco (nr.1). At v = 0.4 m/s (top diagrams) the ranking of this lubricant changes from the fifth (and last) to a shared third position. At 4 m/s the effect of running-in is even more pronounced, as now the ranking changes from the fifth to a first position, shared with the other two mineral oils.



<u>Fig. 4</u> Transition pressures $\bar{p}_c$, at T = 60°C and H = 8 GPa.

It has been shown previously [5, 6, 7] that the I-II and I-III transitions (i.e. the transitions, associated with collapse of EHD lubrication) occur at well-defined values of nominal contact pressure $\bar{p}$. Fig. 4 shows these 'collapse pressures' $\bar{p}_c$ for virginal ($d^0 = 0$) and run-in ($d^0 = 100$ m) surfaces. For virginal surfaces $\bar{p}_c$ has been calculated with the appropriate Hertz' equations; for run-in surfaces $\bar{p}_c$ has been found by dividing the transition force F_c by the area A^0 of the circular wearscar, formed during running-in (i.e. $A^0 = \frac{\Pi}{4} D_w^{0\ 2}$).

It can be seen that - invariably - running-in leads to a reduction in the $\bar{p}_c$ values, thus calculated. At v = 0.4 m/s these reductions range from 5 % (oil 5) to 28 % (oil 3). At v = 4 m/s much higher reductions are found, ranging from 55 % (oil 2) to 80 % (oil 4). The nature of the oil (i.e. mineral or synthetic) does not significantly influences this effect.
The explanation of this reduction in collapse pressure is probably that, even although the running-in process proceeds extremely mild, it produces a process roughness that is much higher than the original roughness of the virginal surfaces (ball bearing components with R_a values of, respectively, 0.01 and 0.12 um c.l.a.). Such an increase in roughness unfavourably affects the load carrying capacity of the partial elastohydrodynamic lubricant film [1]. The fact that, notwithstanding this reduction in collapse pressure values, the corresponding F_c values invariably increase as a result of running-in, is almost certainly due to the formation of a small conformal contact surface (with diameter D_w) during running-in. In other words: mild running-in wear (c.f. column 3 of Table 3) is responsible for the improvement in load carrying capacity (F_c-values).

Contrary to the above, the F_c values at the II-III transition (Fig. 3; middle diagram) can be seen to be virtually independent of running-in and thus, of the formation of a conformal contact area, prior to test. Quite likely this has to do with the fact that in regime II fluid film lubrication effects are completely absent, all force is transmitted by solid-to-solid contacts and the transition to regime III ('unlubricated sliding') is temperature controlled (with critical temperature T_c), rather than pressure controlled.

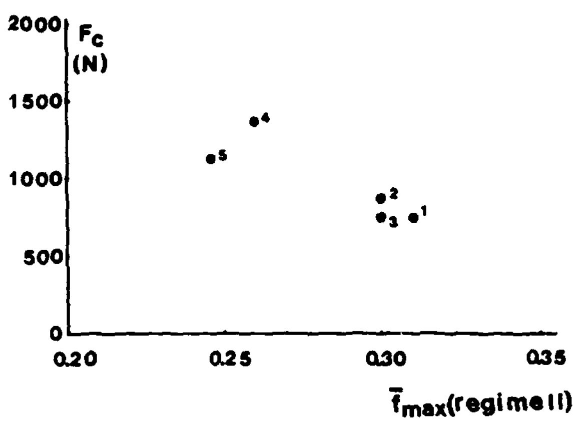


Fig. 5 Relation between the transition force for the II-III transition and the maximum coefficient of friction in regime II.

5.2.2 Effect of lubricant composition

When using virginal surfaces at v = 0.4 m/s (Fig. 3; top left), Royco (nr. 5) produces a much higher load carrying capacity than the other oils, Amoco (nr. 1) clearly being the lastcomer. Running-in (Fig. 3; top right) greatly reduces the differences, although Royco is still in the lead. The relatively good performance of the synthetic oils is all the more remarkable as their viscosity (and - with that - their capacity for lubricant film formation) is much lower than that of the mineral oils. Probably the synthetic oils react more actively with the ball bearing steel than the mineral oils do, thus protecting more effectively the asperities, which - at t = 0 - penetrate the EHD film.

At v = 4 m/s (Fig. 3; bottom diagrams), the synthetic oils clearly fall short in quality, particularly if run-in surfaces are used. Probably at this relatively high sliding speed the supposedly beneficial reactivity of the synthetic oils is more than counterbalanced by their lower viscosity.

Finally at the II-III transition (Fig. 3; middle diagrams) use of synthetic oils clearly has a very beneficial effect. Measurement of friction (see below for friction-time diagrams) shows that this is probably due to the fact that - directly after application of normal force - friction (and thereby contact temperature) is substantially lower with synthetic oils than in the presence of mineral oils or, in other words: the lower the friction, the more force it takes to reach the critical temperature T_c for breakdown of boundary lubrication. Fig. 5 shows the relation between $F_{c_{II-III}}$ and the average value of the maximum coefficient of friction f_{max} in regime II, found experimentally in determining the value of F_c (usually 2 - 3 tests are performed at $F < F_c$).

5.2.3 Correlations among F_c-values

In general Fig. 3 permits the conclusion that the quality ranking of different lubricants, expressed in terms of transition force F_c values, strongly depends on the test results. This is in line with the arguments, put forward in ref. 8.

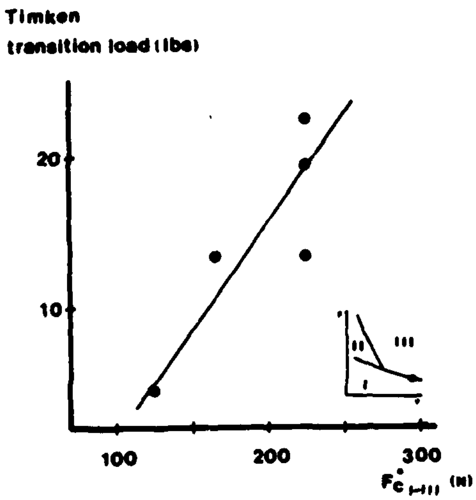


Fig. 6 Correlation between Timken transition load (F_T) and $F^*_{c_{I-III}}$.

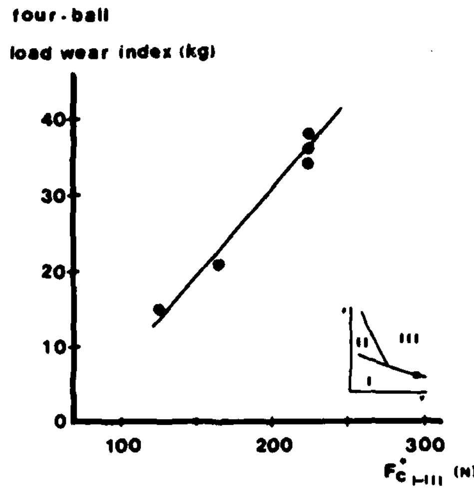


Fig. 7 Correlation between four-ball load wear index (LWI) and $F^*_{c_{I-III}}$.

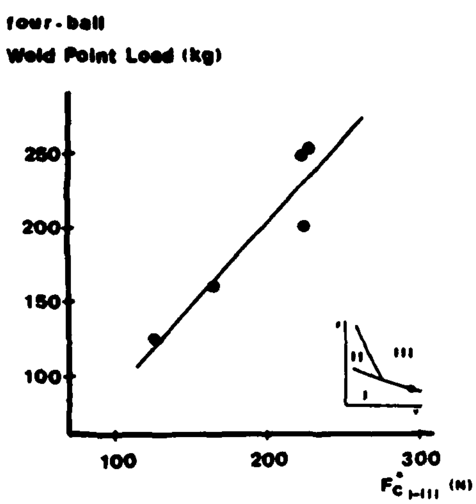


Fig. 8 Correlation between four-ball weld point load (WPL) and $F^*_{c_{I-III}}$.

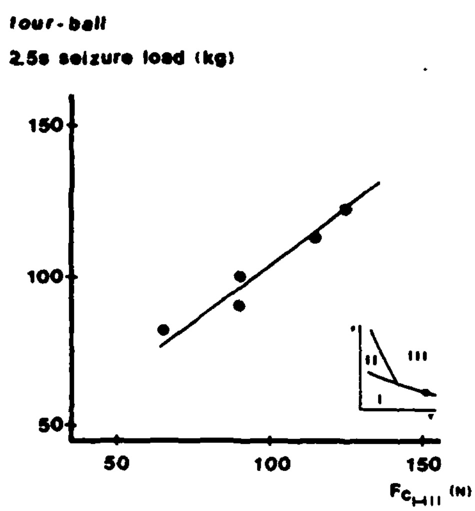


Fig. 9 Correlation between four-ball 2.5 s seizure load ($F_{c_{2.5}}$) and $F_{c_{I-III}}$.

Table 5 shows that a highly significant ($\gamma = 0.97$) positive correlation exists between the $F_{c_{II-III}}$ and the $F^*_{c_{II-III}}$ values, while the $F_{c_{II-III}}$ and $F^*_{c_{II-III}}$ values correlate negatively with the $F^*_{c_{I-III}}$ values. These results completely agree with the information given in par. 5.2.2: at v = 4 m/s the low viscosity of the synthetic oils causes the reverse in quality (c.f. Fig. 3).

In the present case the results of ASTM standard tests (i.e. Timken O.K. and Fail Loads, F_T and four-ball Load Wear Index LWI and Weld Point Load WPL(A) determinations), performed at the U.S. Army Fuels and Lubricants Research Laboratory, were available for comparison. In addition four-ball 2.5 s Seizure loads $F_{c_{2.5}}$ and Weld Point loads WPL(D) were determined at the Laboratory for Machine Components and Tribology of Delft University of Technology (tests performed by Ing. A.J. Hoevenaar). The results of correlation analyses are given in Table 6 in terms of correlation coefficients γ. Table 6 shows that reasonably good correlations exist between F_T, LWI, WPL(A) and WPL(D) values, γ-values ranging from 0.75 (F_T-WPL(A) correlation) to 0.96 (LWI-WPL(A) correlation). On the contrary the $F_{c_{2.5}}$ values do not at all correlate with the other Timken or four-ball results (γ-values range from -0.20 to +0.21). This may have to do with the fact that, upon establishing F_T, LWI or WPL data in 60 s tests, a considerable amount of running-in is involved, while in $F_{c_{2.5}}$ determinations (lasting only 2.5 s) running-in hardly plays a rôle. This point should be elaborated further in discussions among experts and, possibly, future research (see also point c).

5.3.2 Correlations between F_c-values and ASTM standard test data

Table 7 and Figs. 6-9 show that significant ($|\gamma| > 0.90$) correlations between F_c-values and ASTM standard test data occur. Highly positive correlations ($\gamma > 0.90$) between the $F_{c_{I-III}}$ and $F_{c_{2.5}}$-values and between the $F^*_{c_{I-III}}$ and (LWI) and WPL(A)-values are found, while the $F^*_{c_{I-III}}$-F_T and $F^*_{c_{I-III}}$-WPL(D) correlations have relatively high γ-values as well (i.e. 0.86

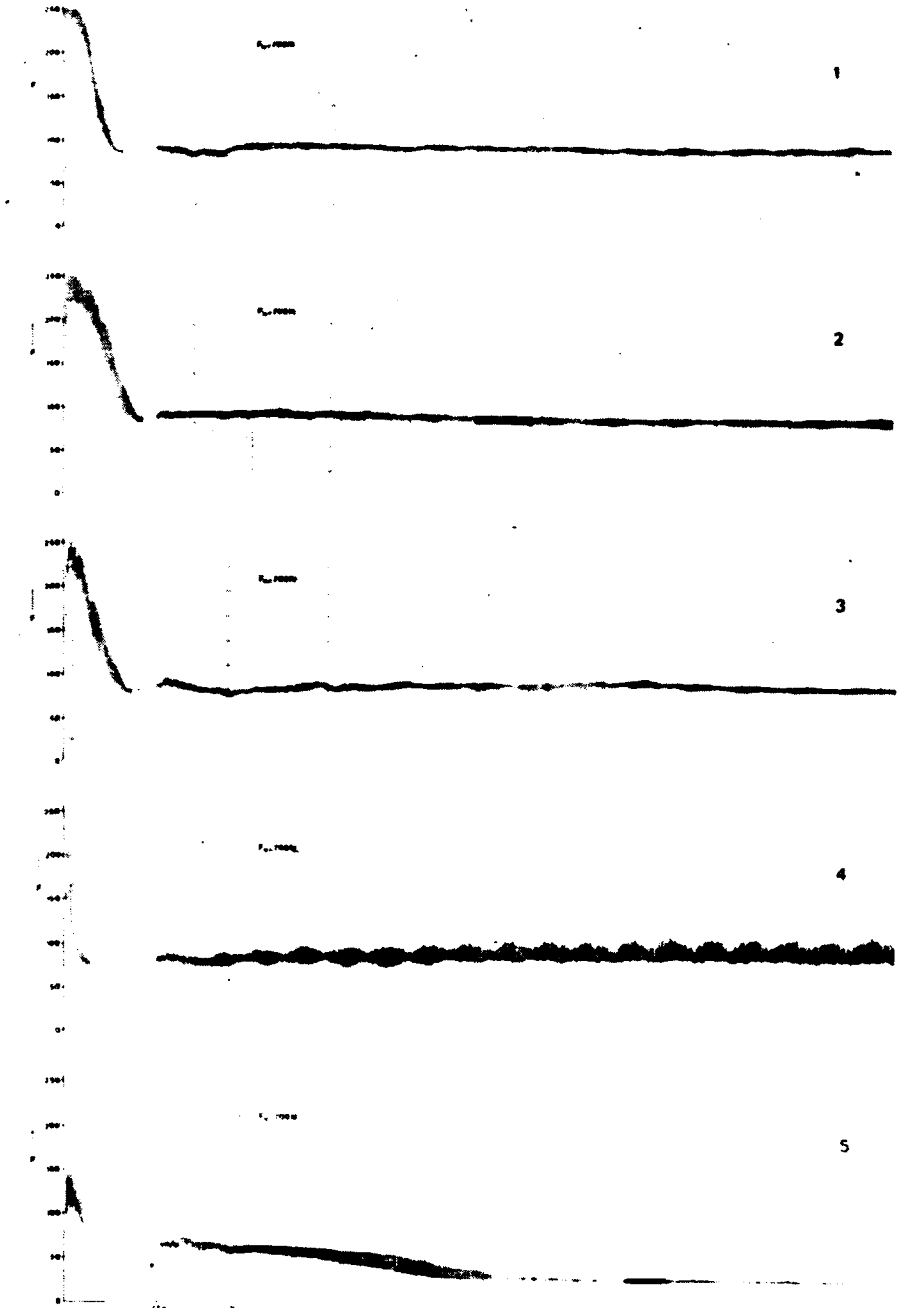


Fig.10 Results of long-term (10 h) friction force F_F versus time t recordings.

and 0.78, respectively). As $F_{c_{I-III}}$ relates to as-machined surfaces and $F^{*}_{c_{I-III}}$ to run-in surfaces, this strongly supports the assumption, put forward in par. 5.3, that $F_{c_{2.5}}$-values relate to as-machined surfaces and LWI and WPL(A)-values to run-in surfaces.

Highly negative correlations ($\gamma > 0.90$) occur between the $F_{c_{II-III}}$ or $F^{*}_{c_{II-III}}$ values and the LWI and WPL(A) values. This comes up to expectations as $F_{c_{II-III}}$ (or $F^{*}_{c_{II-III}}$)values correlate highly negatively with $F^{*}_{c_{I-III}}$ values (see par. 5.2.3).

Remarkably enough, negative correlations are also found for the $F_{c_{I-II}}$ values and the $F_{c_{2.5}}$ values (Table 7: $\gamma = -0.18$) and the $F^{*}_{c_{I-II}}$ and F_T, LWI, WPL(A) and WPL(D) values (γ ranging from -0.41 to +0.77). **This probably identifies the F_T, LWI and WPL data as relating to the I-III transition rather than to the I-II transition.**

Again this point has to be elaborated in future research.

5.4 Friction and wear in regime II of the transition diagram

5.4.1 Friction-time and wear-time curves

An important quality criterion for lubricants is their wear mitigating effect, when functioning under conditions of boundary lubrication (i.e. in lubrication regime II; see Fig. 1). As has been pointed out already, in comparing different lubricants with respect to this property, one has to make sure that, at the beginning of each test, true boundary lubrication conditions prevail in all cases. Table 4 shows that in the present case this requirement can be met at normal forces F between 625 N and 750 N: at F < 625 N lubricant 5 will operate in regime I (very small wearrate); at F_N > 750 N lubricants 1 and 3 will operate in regime III (extremely high wearrate).

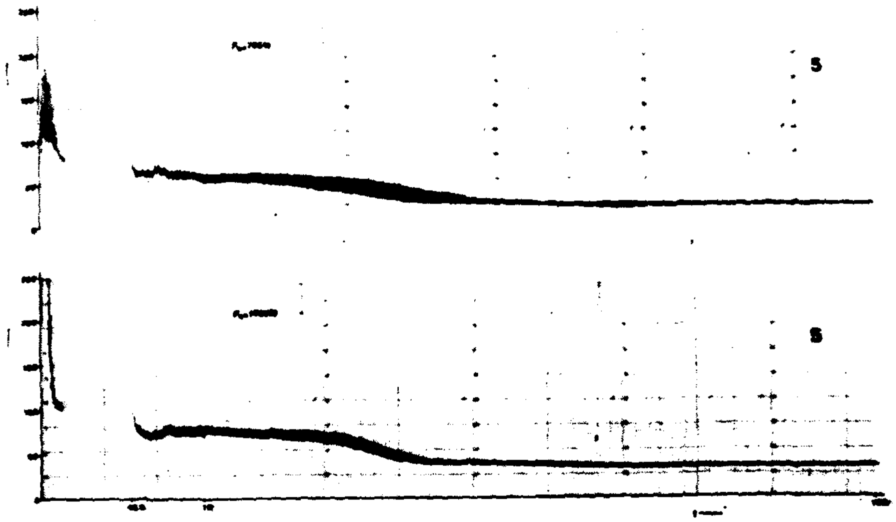


Fig.11 Results of long-term (10 h) friction force F_p versus time t recordings for oil 5 at two different values of normal force F.

The results of long-term (10 h) friction force F_F - time t measurements at v = 0.4 m/s (total sliding distance d = 15 km) with the five different lubricants are shown in Fig. 10 (mark the different time scale at the beginning of each run). Firstly Fig. 10 shows that a marked similarity exists between the F_F/t curves, obtained with the mineral oils (1, 2 and 3), be it that the curves differ in minor details. However, the synthetic oils (4 and 5) behave quite differently, when compared with the mineral oils and when compared among themselves. The first thing that meets the eye is that the maximum friction, recorded some 5 s after application of normal force (as well as the duration of the high-friction period), is much smaller for the synthetic oils than for the mineral oils. This has already been shown in Fig. 5. Fig. 10 further shows that, after reaching the low friction regime, appreciable differences between the different oils emerge. The three mineral oils behave more or less the same, in that early in the contact process the coefficient of friction f (i.e. F_F/F) reaches a rather constant value of about 0.11. When using Mobil (nr. 4) the F_F/t tracing becomes quite 'rough', and at t > 2.5 h (sliding distance d > 3.75 km) high 'friction spikes' occur. These are indicative of occasional metal-to-metal contact. On the contrary with Royco (nr. 5) the F_F/t tracing becomes quite smooth and, in due course, the friction force attains a low value, corresponding with a coefficient of friction f of about 0.035. Clearly in this case full EHD conditions are re-established. As a normal force F of 700 N is only just above the F_c value for the I-II transition for oil 5 (i.e. F_c = 625 N), an additional test with F = 1000 N was performed with this oil. Fig. 11 shows that this does not lead to a significant difference in behaviour, from which it is concluded that the development of friction as a function of time (which is very favourable for a test started under regime II conditions) is indeed characteristic for Royco.

Fig. 12 shows characteristic wear-distance curves, measured in regime II with oils 1-5. When comparing these results with those, shown in Fig. 10, it emerges that the coefficient of friction clearly correlates with the wear rate. Because of the relatively mild 'high friction period' at the beginning of the test, the wear, measured 45 s after application of normal force (i.e. in Fig. 12 at t = 0), is much lower for the surfaces, lubricated with the

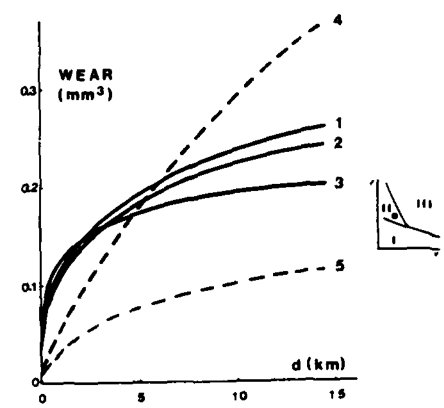


Fig.12 Wear-distance curves, measured in regime II.

synthetic oils (4 and 5) than for surfaces, lubricated with the mineral oils (1 to 3). With Amoco, Citgo, REO and Royco the wearrate decreases appreciably with time (or sliding distance), Royco ultimately 'producing' about half the amount, produced by the other three. With Mobil on the other hand the wearrate hardly reduces at all, in agreement with the irregular friction tracing. The reciprocates of $\frac{d}{\Delta V}$, measured at the end of the tests (i.e. the 'wear resistances') rank in the order Royco, REO, Citgo, Amoco, Mobil. This ranking is different from any of the others, produced during this study (i.e. compare with Figs. 3 and 4).

5.5 Correlation of wear data with Vickers Vane Pump results

For the tree mineral oils, Vickers Vane Pump Wear data were collected at the U.S. Army Fuels and Lubricants Laboratory. These data are listed in Table 8, together with the ΔV^0 values, measured after running-in in regime I (Table 3) and the ΔV values, measured after 15 km running in regime II (Fig. 12). Clearly one does not need correlation analysis to see that correlations do not occur. For the ΔV^0 and ΔV_{15} data this lack of correlation is undoubtedly due to the fact that ΔV^0 and ΔV_{15} were measured in different lubrication regimes and after very different sliding distances (i.e. 0.1 km against 15 km). The total lack of correlation with the Vane pump results is remarkable. In fact one wonders if the very high ring wear (54.2 mg), found with Citgo (nr. 2), is not an artefact. Still the composition of vane and ring materials probably differs considerably from that of ball bearing steel EN31, which may explain the apparent anomaly (c.f. [8]).

5.6 Summary of results, obtained at $T = 60^0$ and $H = 8GPa$

In summary, the results in terms of quality ranking are listed in Table 9. This Table shows that the quality ranking, obtained upon testing five different lubricants, strongly depends on the test conditions, in particular as far as the difference between mineral and synthetic lubricants is concerned. Correlation with (ASTM) standard tests may or may not be found, again depending on the test conditions.

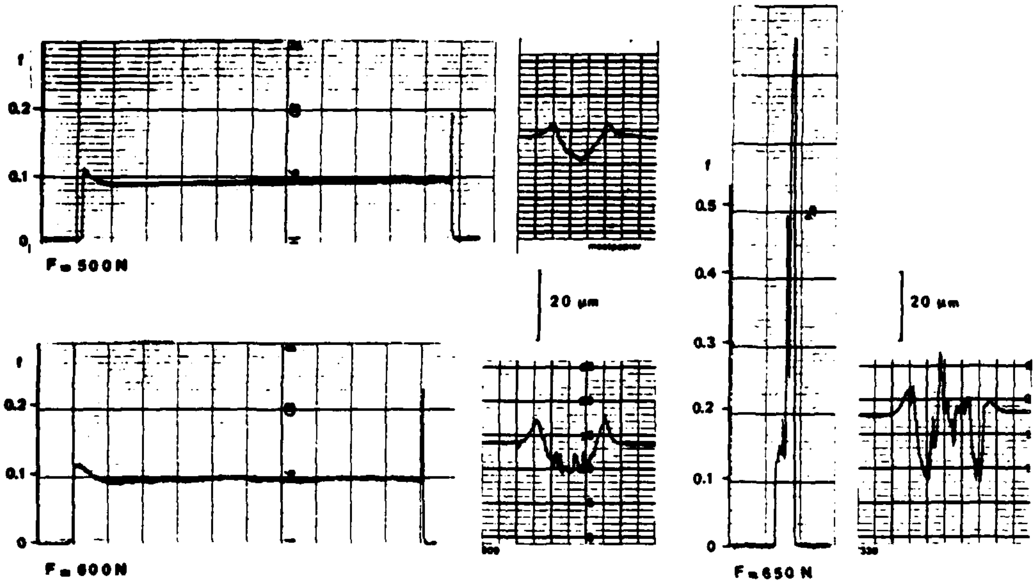


Fig.13 Profile (Talysurf) tracings of ring surfaces, obtained after termination of two minutes' tests and coefficient of friction f versus time recordings.

Lubricant : Mobil (nr. 5)
v : 0.4 m/s
F : 500, 600 and 650 N
H : 3 GPa

6 THE EFFECT OF STEEL HARDNESS AT T = 60°C (test series 10, 11, 14, 15 and 16)

6.1 Morphology of wear tracks and f-t diagrams

When using hard (H = 8 GPa) contact surfaces, the transition pressures (c.f. Fig. 4) remain below the critical pressure p^* for beginning plastic deformation ($p^* \approx \frac{1}{3} H \approx 2.8$ GPa). However, upon using relatively soft (H = 3 GPa) contact surfaces and under conditions of dynamic contact with traction (f ≈ 0.1) a considerable amount of plastic deformation occurs. This can be seen from Fig. 13, which shows profile (Talysurf) tracings of ring-surfaces, measured after termination of two min. tests with Mobil (nr. 4) at v = 0.4 m/s and under normal forces F of 500. 600 and 650 N, respectively. It can also be seen that at F = 500 N the tracing is very smooth (irregularities of the order of 1 um). At F = 600 N, roughness peaks with a height of the order of 5 um appear at the bottom of the groove and, finally, at F = 700 N, the groove pattern has all but disappeared, due to considerable 'secundary roughening effects'. Microscopic observation of the wear tracks reveals that such roughening is due to severe adhesive steel transfer from the ball to the ring surface.

The three profile tracings already suggest that a transition occurs at a force F between 600 and 700 N, i.e. probably close to F = 600 N. The same is borne out by the coefficient of friction f-time t tracings, which are also shown in Fig. 13. At F = 500 N and F = 600 N the coefficient of friction f stays at the 0.1 level, characteristic for lubrication regime I. At F = 650 N, however, f rises to extremely high values, within a few seconds after the application of normal force F. The profile tracings as well as the f-t recordings suggest that (at v = 0.4 m/s!) a I-III transition occurred. This was confirmed by measuring wear at F = 650 N: very severe adhesive wear of the ball occurred (c.f. par. 2.1, page 11).

Performing all the individual tests of test series 10, 11, 14, 15 and 16 showed that the above behaviour is characteristic for that of all five lubricants.
Thus at v = 4 m/s as well as at v = 0.4 m/s I-III transitions occurred.

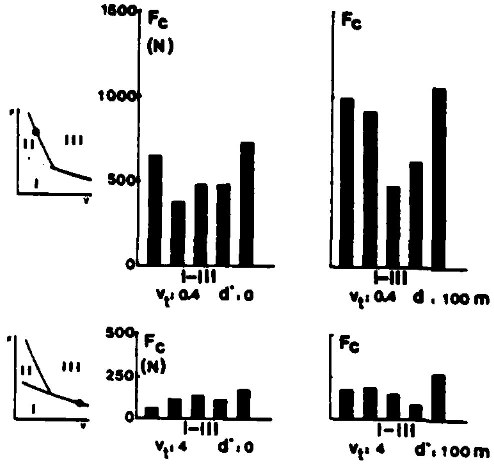


Fig.14 Transition forces F_c at $T = 60^{\circ}C$ and $H = 3$ GPa.

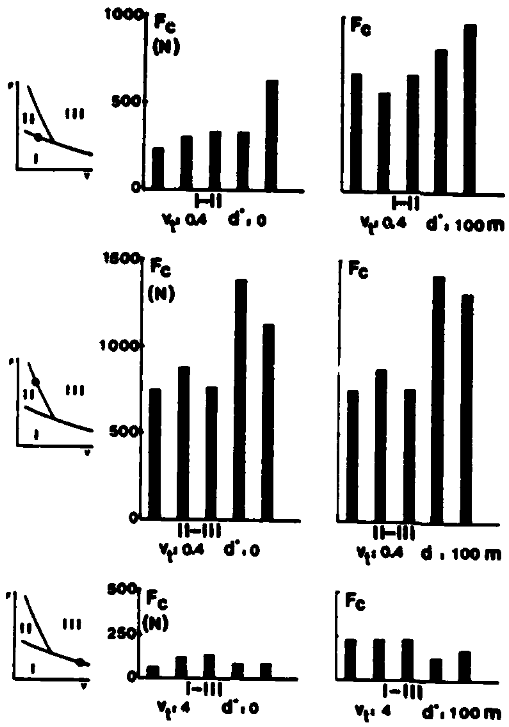


Fig.3 (from page 16)

6.2 Transition forces

Transition forces, determined with an accuracy of $\pm$ 25 N,are given in Table 10 and Fig. 14. To facilitate comparison with the results obtained with hard (H = 8 GPa) steel surfaces, Fig. 3 is reproduced on the same page. Looking firstly at the results, produced at v = 4 m/s, it can be seen that for lubricants 1 to 4 the enormous difference in hardness and deformation behaviour of the contact surfaces has remarkably little effect on F_c. However, Royco oil (nr. 5) behaves much better in combination with 'soft surfaces', than in combination with 'hard surfaces'.

As stated above, when using soft contact surfaces, a I-III transition also occurs at v = 0.4 m/s. This results in rankings which are distinctly different from those found with hard contact surfaces (upper four diagrams of Fig. 3). Remarkable details are the relatively good performance of Amoco (nr. 1) and the disappointing behaviour of Mobil (nr. 4).

6.3 Summary of results, obtained at T = 60^o and H = 3 GPa

Using relatively soft contact surfaces leads to severe plastic deformation, prior to transition. Invariably, I-III transitions occur, i.e. also at v = 0.4 m/s. At v = 0.4 m/s and v = 4 m/s different rankings are obtained, which also differ from any of the rankings produced before (Table 9). Under conditions of severe plastic deformation Royco behaves exceptionally well.

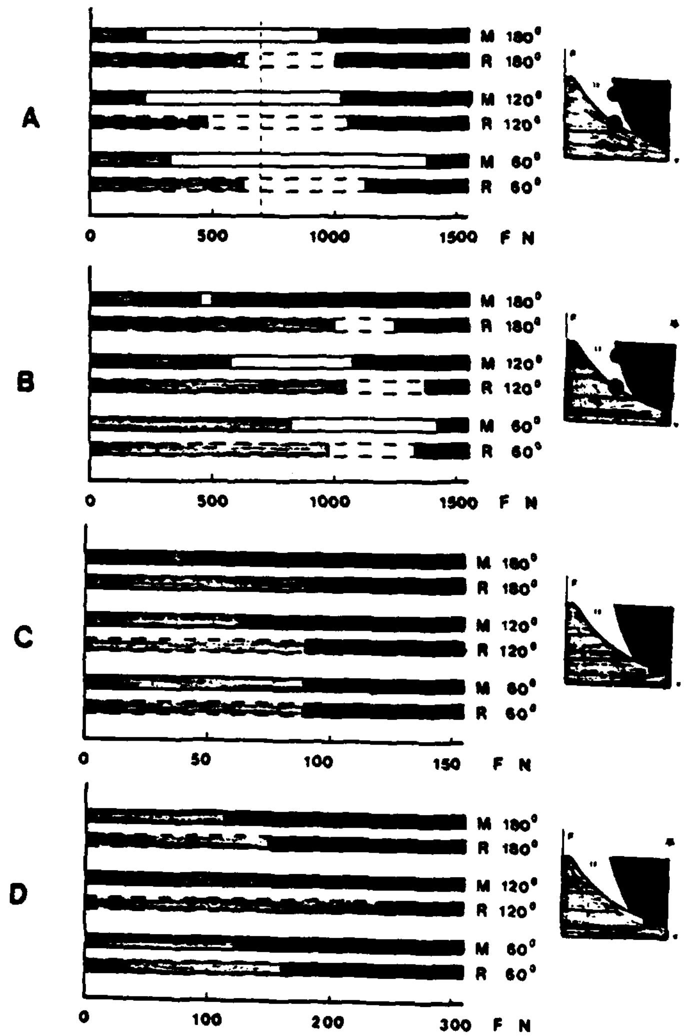


Fig.15 Results of F_c-determinations (from Table 11).

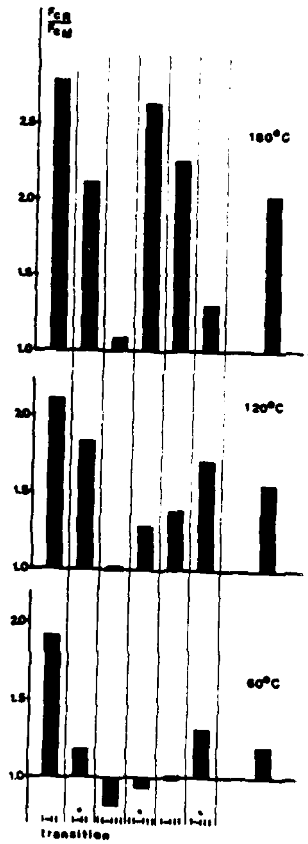


Fig.16 Ratios F_c Royco/F_c Mobil for the parameter combinations, given in Table 11 (data from Table 12).

7 THE EFFECT OF LUBRICANT BATH TEMPERATURE (test series 6 to 9, 12, 13 and 17)

7.1 Mobil and Royco at H = 8 GPa

With the exception of Amoco at T = 120°C (test series 17), the tests at oil bath temperatures over 60°C relate to the two synthetic lubricants Mobil and Royco. In order to facilitate comparison, the results obtained with these lubricants at H = 8 GPa and T = 60°C (test series 4 and 5, already listed in Table 4), T = 120°C (test series 6 and 7) and T = 180°C (test series 8 and 9) are discussed together.

7.1.1 Transition forces

Transition forces F_c, determined at v = 0.4 m/s and v = 4 m/s, are shown in Table 11 and Fig. 15. From these data it can be seen that for Mobil, with the exception of the F^*_{cI-III} results, an increase in oilbath temperature causes a gradual decrease in F_c-value. As a result the average F_c-value ($\bar{F}_c$) (averaged over the seven entries on one line of table 11) decreases with increasing oilbath temperature in the sequence: 595 N, 444 N, 322 N. With Royco a similar systematic effect does not occur, $\bar{F}_c$-values at T = 60, 120 and 180°C being, respectively, 615 N, 611 N and 588 N.

The differences between the two types of synthetic oil can be brought out more clearly by calculating the ratios F_{cRoyco}/F_{cMobil}. The values of these ratios are listed in Table 12 and Fig. 16.

Table 12 and Fig. 16 show that Royco oil generally provides a higher load carrying capacity than Mobil oil, the only exceptions being the data, relating to the II-III transitions with as-machined and with run-in surfaces, determined at 60°C oil bath temperature.Detailed conclusions from Table 12 are:

Individual ratios:
Largest ratio (2.78): I-II transition at 180°C.

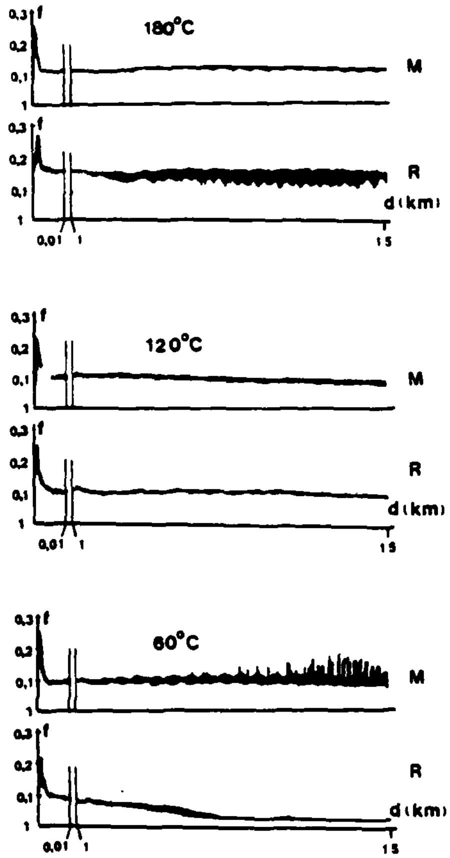


Fig.17 Coefficient of friction f is a function of sliding distance d. Lubricants: Mobil (M) and Royco (R)
T = 60, 120 and 180°C
V = 0.4 m/s
F = 700 N

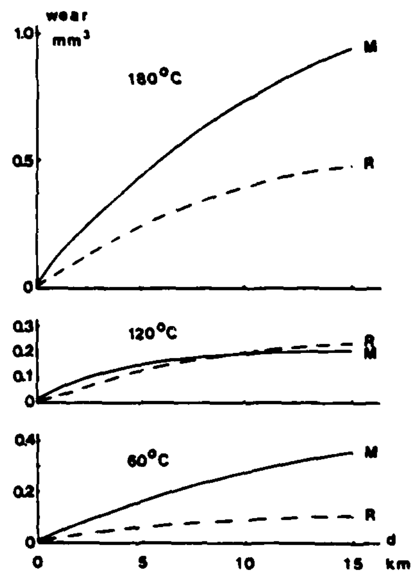


Fig.18 Volume wear as a function of sliding distance d.
Lubricants: Mobil (M) and Royco (R).
T = 60, 120 and 180°C
V = 0.4 m/s
F = 700 N

Smallest ratio (0.82): II-III transition at $60^\circ C$.

Average values all transitions:
Ratio increases significantly with increasing temperature.

Average values all temperatures:
Largest ratio (2.27): I-II transition.
Smallest ratio (0.97): II-III transition.

Total average:
Ratio = 1.59, i.e.: on the average Royco provides 59% more load carrying capacity than Mobil.

7.1.2 Friction and wear in region II

Coefficient of friction f versus sliding distance d diagrams for tests, started in lubrication regime II (boundary lubrication) are shown in Fig. 17.
The tests were performed at $v = 0.4$ m/s and $F = 700$ N. Fig. 3, top diagram, shows that at this combination of v and F true boundary lubrication in indeed established in all cases. Fig. 17 shows that increasing the oil bath temperature from $60^\circ C$ to $120^\circ C$ eliminates the difference in friction behaviour between Royco oil and Mobil Oil, that in observed at $60^\circ C$.
A further increase of oil bath temperature to $180^\circ C$ results for both oils in a significant increase in friction (i.e. from $f \approx 0.12$ to $f \approx 0.15$). Also the f-d tracing for Royco becomes 'rough'. Nevertheless this tracing looks better than that found with Mobil oil at $60^\circ C$ (no 'spikes'; i.e. probably no unprotected metal-to-metal contacts).
The wear behaviour of the two oils at 60. 120 and $180^\circ C$ is shown in Fig. 18. This Fig. shows that, surprisingly, at $180^\circ C$ Royco oil again performs better than Mobil oil.

The amount of volume wear, measured at $d = 15$ km, is shown seperately in Fig. 19.

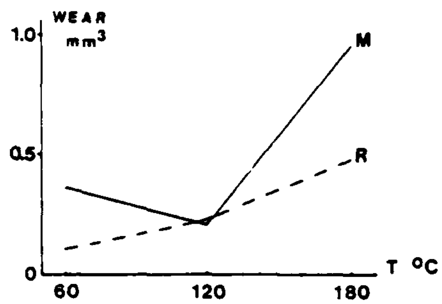


Fig.19 Volume wear at sliding distance d = 15 km as a function of oilbath temperature T.
Lubricants : Mobil (M) and Royco (R).
Test conditions: see caption Fig. 18.

Fig. 18 shows that the amount of volume wear of Mobil oil goes through a minimum, while that of Royco oil shows a steady increase with increasing temperature.

This may be explained by assuming that Mobil oil contains an additive, that needs temperature (i.e. activation energy) to become active, but which desorps (or decomposes) at still higher temperatures.

7.2 Mobil and Royco at H = 3 GPa (test series 12 and 13)

Both Mobil and Royco have been tested at T = 180°C and H = 3 GPa. Again, severe plastic deformation took place and I-III transitions occurred at v = 4 m/s as well as at v = 0.4 m/s. Table 13 contains the results in terms F_c values. To facilitate comparison, the results obtained at T = 60°C and H = 3 GPa (already listed in Table 10) are given as well. It can be seen that with Mobil an increase in T from 60° to 180°C course a pronounced decrease in F_c. This is in line with the results given in Table 11. With Royco, however, a similar increase in T causes a remarkable **increase** in F_c-values! Actually at v = 0.4 m/s as well as at v = 4 m/s the relevant F_c-values are by far the highest ever found in the course of this research program. **This identifies Royco as a lubricant, highly suitable for use under conditions of severe plastic deformation at high temperature.**

7.3 Amoco at H = GPa and T = 120°C (test series 17)

Table 14 shows the results of testing Amoco at T = 120°C, using specimens with H = 8 GPa. For comparison the results obtained at 60°C are shown as well (already shown in Table 4). It can be seen that all F_c-values decrease as a result of increase in oilbath temperature. Thus in this respect Amoco behaves similar to Mobil (but quite different from Royco!).

7.4 Summary of results, obtained at different lubricant bath temperatures

With Mobil (tested at T = 60, 120 and 180°C) and Amoco (tested at T = 60 and 120°C) the load carrying capacity F_c generally decreases with increasing oil bath temperature T. On the contrary the F_c-values of surfaces, lubricated with Royco (tested at T = 60, 120 and 180°C), remain roughly the same (at H = 8 GPa) or increase appreciably with increasing T (at H = 3 GPa). Citgo and REO were not tested at elevated oilbath temperature.

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Table 1 Lubricants tested

Oil			ν 60°C
nr	designation	type	$mm^2 s^{-1}$
1	Amoco 300 Motor Oil 15W-40 (MC - 2531)	mineral	44
2	Citgo C-5 Motor Oil 15W-40 (MC - 2512)	mineral	46
3	REO 203 Motor Oil Grade 30	mineral	44
4	Mobil Jet II	synthetic	15
5	Royco 555	synthetic	14

Table 2 Survey of test program

The numbers indicate the order in which the tests were performed.

Test condition	Lubricant				
	Amoco	Citgo	REO	Mobil	Royco
T = 60°C H = 8 GPa	1	2	3	4	5
T = 120°C H = 8 GPa	17	-	-	6	7
T = 180°C H = 8 GPa	-	-	-	8	9
T = 60°C H = 3 GPa	14	15	16	10	11
T = 120°C H = 3 GPa	-	-	-	-	-
T = 180°C H = 3 GPa	-	-	-	12	13

Test series 1-5: results also reported in the second interim report (also called Anual Technical Report) of June 1986.

Test series 6-9: results also reported in the fourth interim report of August 1987.

Test series 10-17: not previously reported.

Table 3 Wear during running-in and D_E values

oil	running-in		test		
			$v = 0.4\ ms^{-1}$		$v = 4\ ms^{-1}$
			I-II	II-III	I-III
	D_w^0	ΔV^0	D_E	D_E	D_E
	um	$10^{-3}\ mm^3$	um	um	um
1	750	2.8	560	580	390
2	650	1.6	540	600	390
3	750	2.8	560	580	390
4	775	3.1	600	720	320
5	700	2.1	640	700	360

D_w^0 = Diameter of wearscar, formed during running-in

ΔV = running-in wear

D_E = Diameter of circular contact surface, which would form on virginal test surfaces by elastic deformation at $F = F_c$ (F_c values in Table 4).

Table 4 Transition force data and data from other sources
$T = 60^{0}C$, $H = 8$ GPa

Lubricant	TNO						Army			TUD	
	V = 0.4 m/s				v = 4 m/s						
	F_c I-II	F_c^* I-II	F_c II-III	F_c^* II-III	F_c I-III	F_c^* I-III	F_T	LWI	WPL	$F_{c2.5}$	WPL
	N	N	N	N	N	N	lbs		kg	kg	kg
1. Amoco	225	675	750	750	65	225	22.5	38	250	82	300
2. Citgo	300	575	875	875	115	225	19.5	34	200	113	200
3. REO	325	675	750	750	125	225	13.5	36	250	123	250
4. Mobil	325	825	1375	1425	90	125	4.5	15	126	100	150
5. Royco	625	975	1125	1325	90	165	13.5	21	160	90	200

F_c = load carrying capacity (transition load) of EHD or boundary film (* : run-in surfaces).

F_T = average between Timken OK and fail loads.

LWI = four-ball load wear index.

WPL = four-ball weld point load

$F_{c_{2.5}}$ = four-ball 2.5 s seizure load

TNO = Metals Research Institute TNO data

Army = data, supplied by U.S. Army

TUD = data, supplied by Technological University Delft.

Table 5 Results of correlation analyses; γ-values: correlations among F_c-values.

	F_c I−II	F_c^* I−II	F_c II−III	F_c^* II−III	F_c I−III
F_c^* I−II	83				
F_c II−III	42	68			
F_c^* II−III	61	81	97		
F_c I−III	4	-33	-19	-21	
F_c^* I−III	-44	-76	-98	-97	25

Table 6 Results of correlation analyses; γ-values: correlations among ASTM standard test data.

	F_T	LWI	WPL A	WPL D
LWI	85			
WPL A	75	96		
WPL D	78	85	92	
$F_{C_{2.5}}$	-20	21	18	-21

Table 7 Results of correlation analyses; γ-values: correlations between F_c-values and ASTM standard test data.

	F_T	LWI	WPL A	WPL D	$F_{C_{2.5}}$
F_{C} I – II	-30	-54	-49	-39	-18
F_{C}^{*} I – II	-55	-77	-64	-41	-47
F_{C} II – III	-83	**-99**	**-98**	-87	-22
F_{C}^{*} II – III	-78	**-99**	**-97**	-83	-29
F_{C} I – III	-18	16	12	-27	**97**
F_{C}^{*} I – III	86	**99**	**93**	78	28

Table 8 Summary of wear results

Lubricant	ΔV^0 1) 10^{-3} mm^3	ΔV_{15} 2) 10^{-2} mm^3	Vickers Vane Pump Wear (weight loss in mg)		
			Ring	Vane	Total
1. Amoco	2.8	26	17.4	5.0	22.4
2. Citgo	1.6	24	54.2	7.3	61.5
3. REO	2.8	20	17.5	2.3	19.8
4. Mobil	3.1	37	n.t 3)	n.t	n.t
5. Royco	2.1	11	n.t	n.t	n.t

1) From Table 3

2) From Fig. 12

3) n.t = not tested

Table 9 Quality ranking of lubricants

Lubricant	transition I-II		transition II-III		transition I-III		wearresistance
	no run-in	run-in	no run-in	run-in	no run-in*	run-in**	in regime II
1 Amoco	4	3	4	4	4	1	4
2 Citgo	3	4	3	3	2	1	3
3 REO	2	3	4	4	1	1	2
4 Mobil	2	2	1	1	3	3	5
5 Royco	1	1	2	2	3	2	1

1 = best quality

5 = least quality

*) This ranking correlates well with that of four-ball 2.5 seizure load ($F_{c2.5}$) data.

**) This ranking correlates well with that of Timken O.K. load (F_T) and four-ball Load Wear Index (LWI) and Weld Point Load (WPL) determinations.

Table 10 Transition force data, obtained at T = 60°C and H = 3 GPa

Lubricant	v = 0.4 m/s		v = 4 m/s	
	F_c I-II	F_c^* II-III	F_c^* II-III	F_c^* I-III
	N	N	N	N
1. Amoco	650	1000	65	175
2. Citgo	375	925	115	190
3. REO	475	475	140	150
4. Mobil	475	625	115	90
5. Royco	825	1075	175	275

Table 11 Transition forces F_c
T = 60, 120 and 180° C   H = 8 PGa

Lubricant	V = 0.4 m/s				v= 4 m/s		$\overline{F}_c$
	F_c I-II N	F_c* I-II N	F_c II-III N	F_c* II-III N	F_c I-III N	F_c* I-III N	N
Mobil 60°	325	825	1375	1425	90	125	595
Royco 60°	625	975	1125	1325	90	165	615
Mobil 120°	225	575	1025	1075	65	140	444
Royco 120°	475	1050	1050	1375	90	240	611
Mobil 180°	225	475	925	475	40	115	322
Royco 180°	625	1000	1000	1250	90	150	588

Table 12 Ratio of load carrying capacities $\frac{F_{c\ Royco}}{F_{c\ Mobil}}$ for the parameter combinations, given in Table 11.

Oil bath temperature	F_c I-II N	F_c* I-II N	F_c II-III N	F_c* II-III N	F_c I-III N	F_c* I-III N	Average
60	1.92	1.18	0.82	0.93	1.00	1.32	1.20
120	2.11	1.83	1.02	1.28	1.38	1.71	1.56
180	2.78	2.11	1.08	2.63	2.25	1.30	2.03
Average	2.27	1.71	0.97	1.61	1.54	1.44	1.59

Table 13 Transition forces F_c, obtained with Mobil and Royco at H = 3 GPa and T = 60 and 180°C

Oilbath T (°C)	Lubricant	V = 0.4 m/s		v = 0.4 m/s	
		F_c I-III N	F_c* I-III N	F_c I-III N	F_c* I-III N
60	Mobil	475	625	115	90
180	Mobil	90	275	65	25
60	Royco	825	1075	175	275
180	Royco	2050	2050	375	375

Table 14 Influence of oil bath temperature T on the F_c-values of Amoco H = 8 GPa.

Lubricant	v = 0.4 m/s				v= 4 m/s	
	F_c I-II N	F_c* I-II N	F_c II-III N	F_c* II-III N	F_c I-III N	F_c* I-III N
1. Amoco 60°	225	675	750	750	65	225
1. Amoco 120°	175	475	575	550	40	175

END

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